LLOYD'S REGISTER OF SHIPPING

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LARGE POLAR ICEBREAKER FOR UNITED STATES COASTGUARD

Preliminary Report on an Engineering Study to determine the Optimum Structure for a large Polar Icebreaker as per Coast Guard Contract Tcg-17961-A of 26th January, 1967.

This report refers to the historical background of icebreaker design, discusses the merits of the theoretical work done and suggests the criteria to be adopted for determining shell plating and framing structure.

Reference is also made to the effect of aspect ratio on shell thickness with corresponding curves. Corrosion of icebreakers is also briefly discussed.

A brief summary of the work presently in progress and recommendations for the conclusion of work are also given.

1. Historical Background

The evolution of the present day icebreaker hull design has been based on practical experience gained in breaking ice. Hull form is extremely important as the impact and icebreaking efficiency of a hull is a direct function of the hull lines. Hydrostatic loads are not generally significant when compared with ice pressures particularly when navigating in polar ice.

In the past empirical methods have been applied to the hull structure with scantlings at the ends of the vessel made heavier than midships. The determination of ice pressure is very difficult. Pressure can vary quite considerably over a field of ice, for ice in a small block may have quite a high compressive strength whereas when the block becomes part of an ice field it may not be of such importance due to the inertia effect of the ice field. For large Polar icebreakers it has been the custom to design the structure to withstand the maximum ice pressure likely to be met in service at the Poles. A large icebreaker will have a greater mass than a smaller icebreaker hence increasing the scantlings of a small icebreaker to that of a large icebreaker will not necessarily mean an increase in icebreaking capability. The first icebreakers were in fact icebreaking tugs operating in thin river ice, and they gradually increased in size with icebreaking requirements and range. In general icebreakers were, and still are to a large extent, built to the requirements of national authorities rather than merchant shipowners and the strength requirements and even the design of hull structure, have not in general followed an overall strength standard as is common in merchant ship construction. It has been the requirements and philosophy of the individual authority that has quite often decided the type and strength of structure required and in general the scantlings have been in excess of those required by the Classification Societies.

This makes comparison difficult between the scantlings of individual icebreakers.

2. Theoretical Approach

Most of the theoretical work done on vessels working in ice has been carried out by Russian engineers and is generally of a highly mathematical content.

References are given at the end of the report to the papers mentioned below and the main purpose of discussing the papers in this report is to highlight the advantages and disadvantages of each paper in turn as they apply to the practical structural design of an icebreaker hull, and to see if use can be made of the theories propounded.

Paper No. 1. "Ice Loads Acting on Ships" by M.K. Tarshis.

This paper sets out to determine the impact load of a vessel hitting the ice and the area of hull in contact.

The basic formula put forward is of the form

$$\text{Impact load} = \text{Speed} \times \text{Angle of blow}\sqrt{\text{Rel.mass and rel. rigidity}}$$

The author specifies the strength of ice in crushing and bending and assumes that the crushing strength of ice is the criteria for determining the load.

It is necessary to know three angles namely

(i) Angle between the tangent to the waterline at the point of impact and the fore and aft plane of the ship;

(ii) Angle between the tangent to the frame at the point of impact and the plane of the frame;

(iii) Angle between the tangent to the buttock line at the point of impact and the plane of the frame.

A numerical example is given in the paper for a small icebreaker of 655 tons displacement in contact with a round floe d = 25 metres and 1 metre thickness. The required information on hull form, speed, draft, etc., and point of contact is given to enable the example to be solved, from which a contact load of 220 tons is deduced. A load area is also deduced, which in this example is given as

1.18 metres thick x 0.46 metres long

An example has been worked by Lloyd's Register using the formulae and assumptions of the author for a large vessel of about the size of "MOSKVA", having a displacement of 15,000 tons hitting the ice at the same speed and same relative point of contact as the numerical example - all other items being unchanged.

It is interesting to see that increasing the displacement by about 23 times only increases the load of impact from 220 tons to 291 tons and the load bearing area now becomes

1.18 metres thick x 0.614 metres long

This is because the basic formulae used by the author does not consider the relative mass of the icebreaker to have as great a bearing on the impact force as the speed and angles of hull form in contact.

Reference to the basic formula shows that doubling the impact speed will double the impact load.

A further example was calculated by Lloyd's Register assuming the "MOSKVA" type dimensions, etc., with the same contact speed and angles, but this time in ice 3 metres thick.

The resulting impact force increased by 2.9 times to 843 tons and the load bearing area became

3.55 metres thick x 0.594 metres long

If the diameter of floe is assumed infinite the impact force increases but as the crushing strength of ice does not increase, the spread of the load increases in length. A check was therefore made of the effect on the impact load for an ice-breaker of the "MOSKVA" size and hitting an infinite floe and the impact load increased as a result by about 2½ times with a corresponding increase in length of spread.

It is claimed that the hardness of the ice does not affect the impact load but rather the area in contact. Hard ice will have a shorter spread than soft ice. Increasing the thickness of the ice increases the actual impact load but not the intensity of load.

The crushing strength of ice used in the examples is 570 lbs/in^2.

By using the assumptions and formulae given it is theoretically possible to determine the impact load and area in contact, from which it can be seen that concentrated, point type loads are not likely to occur, although it is possible for one single frame to be loaded by a hard spot in the ice.

The paper has a high theoretical content and requires many parameters difficult to determine. The relative angles of the hull form at point of impact are necessary and these can vary quite appreciably in a relatively short distance.

The paper is useful, however, in showing that speed and hull form are very important for determining principal hull scantlings and also that point loads need not be taken into account.

Paper No. 2. "Impact of Ships with Ice" by L.M. Nogid

This paper sets out to determine the reduction in speed of an icebreaker of known dimensions, mass, hull form and hull strength at point of impact, assuming two crushing ice strengths and two ice thicknesses.

It is interesting to note that assuming a ratio of ship's mass to floe mass of 20, the speed is about 4 times that when travelling in an infinite floe mass.

In addition the maximum speed varies in direct proportion to the maximum load which the ship's hull can withstand at the point of contact.

The author shows that the force required to crack the ice is theoretically quite small (about 7 tons for an ice thickness of 0.6 metres and a crushing strength of 170 p.s.i.), whereas practical data indicates that a far greater force has to be applied to break the ice field.

The paper goes on to show that theoretically the force required to break off sectors of ice is more than 3 times the force corresponding to the beginning of crack formation.

The paper can be summarized by saying that knowing the maximum load which the ship's hull can withstand at the point of contact, the angle formed by the line of impact and the hull form, then for various ice floe diameters the reduction in speed can be theoretically deduced. The area in contact can also be deduced knowing the crushing strength of the ice.

The author recommends that the scantlings of a frame should be sufficient to withstand the impact force and the external plating should be based on an evenly distributed load with an intensity equal to the crushing strength of the ice.

Again the paper has a highly mathematical content and is difficult to apply in practice. However, it is useful in showing the sort of speed reductions to theoretically expect when operating in ice floes of bigger and bigger dimensions.

Paper No. 3. "Determination and Appraisal of the Structural Strength of Ships Navigating in Ice by Recalculating from the Prototype by D.E. Kheisin.

In this paper it is assumed that all is known

about the prototype, including the lines and the design strength of the structure.

It is important to know how the prototype stands up under actual service conditions and true records must be kept of damages sustained, the angle of such damages, the ice thickness and strength and speed of ship if the methods proposed by the author are to be of any value.

The author states that a ship navigating in ice will be subjected to two types of force irrespective of its type and duty, namely -

1. Impact
2. Compression

The impact loads will determine the necessary strength of the ends of the ship and the compressive force will be the design criteria for the middle part of the ship.

It is assumed that the loads can be such as to cause the stresses in the hull structure to just reach yield point in the particular portion of the hull under consideration.

The strength requirements are also divided into two categories, namely -

1. Impact and 2. Compression

and these can be briefly summarized as follows :-

1. Impact: A basic formula of $S = \frac{M_1 v \ell}{C}$ is propounded where S = impact force
M_1 = mass of ship
v = speed of ship at contact
ℓ & C are parameters depending upon hull form, point of impact, etc.

A further assumption is made that the above formula is for impact with an infinite ice field.

Ice friction, elastic deformation of the ice, and deflection of ice framing are assumed negligible and in thick ice bending of the ice field is ignored.

2. Compression: Here the author assumes that the governing criteria is the crushing strength of the ice. Speed does not enter into the calculations in this instance, and the basic parameters are ice pressure and angle of framing to ice.

It is suggested that length of ship has some bearing on the strength of the side frames and this is due to the fact that bending of ice does occur, the magnitude of which depends upon the L/B ratio, hull form and length of ship. An interesting point brought out by the author is that if a vessel is all parallel middle body, i.e. no flare, length does not have any effect.

The formula suggested by the author to take account of length is

$$\left(\frac{K_o L_o}{K_1 L_1}\right)^{\frac{1}{3}}$$

K_o and K_1 being coefficients of prototype and design depending upon L/B ratio.

Using the above formula, increasing the length by 50% increases the requirement for design strength of the side frames by about 14%.

The above comments refer to the strength of main framing under compression. The author also investigates the hull plating under compression and shows that shell thickness is directly proportional to frame spacing and varies as the square root of the yield point of the material. A further factor is brought in depending upon ice thickness and frame spacing and shows that the wider the ice belt the thicker the shell needs to be with a limiting ice thickness of 1.6 x frame spacing. This is based on the premise that the intensity of loading is proportional to the ultimate strength of ice against crushing.

The author also assumes that the critical stresses occur at the mid span of the plate whose edges are assumed to be rigidly fixed.

In the paper the frame loading is assumed to act in a line uniformly distributed at mid span of frames

and for stringers a uniform line load acts directly on the stringer.

Fig. 6 in the paper shows design strength profiles which clearly indicate that shoulder pressures are very high and need careful attention.

Before full use can be made of this paper it is necessary to have all the facts of the prototype, the damage reports, strength criteria of hull, etc., and as such the paper is not directly applicable when designing a completely new type and size of icebreaker, unless very careful research can be done. The advantage of the paper however is that the author suggests that hull strength in ice should be approached in two ways, viz. Impact and Compression.

The Impact formula propounded is basically of the momentum type depending upon mass and velocity and should be applied to the ends of the vessel and for the middle portion of the ship the structure should be designed on the ultimate strength of the ice against crushing.

Paper No. 4. "Method of Determining the Stresses in Decks and Transverse Bulkheads Caused by Ice Loads" by Yu. N. Raskin

In this paper the author attempts to calculate the stresses in decks and bulkheads under a compressive ice load.

With regard to the decks the author divides the deck into strips with the stringer being in simple compression depending only upon area of plating and beams in contact and compared with the Euler stresses in the stringer including beams in way. It is then assumed that the remaining "bands" of plating act as short beams which are infinitely rigid as regards bending and which work in shear under the action of the ice loads. It is further assumed that between these "bands" there are elastic inserts which allow relative movement between the bands.

This is a theoretical paper making several

assumptions which the author contends are necessary if very laborious calculations are to be avoided which would certainly be necessary if the methods of elastic theory were applied to structures of this type.

It is difficult to visualize how the contents of this paper can be applied in practice to a complicated hull structure of the type normally associated with icebreakers, but nevertheless, it has some usefullness in that it shows that cross sectional area is important and that lengths of panels between supports should not be too long under the action of a compressive ice field.

SUMMARY OF THEORETICAL APPROACH

As can be seen from the above, the Russian work is of a highly theoretical nature with many of the assumptions incalculable, thus making practical application difficult. Despite these comments, however, certain guidance lines emerge and these are briefly summarised as follows.

Icebreakers should be designed from two aspects :-

1. Impact at ends
2. Compression at mid length.

Basically, Impact = Speed x function of mass.

Tarshis brings in angle of blow and hull form at point of impact. He does not place much emphasis on mass of ship and considers that impact force varies as $\sqrt{\text{mass} \times 10^{3}}$. He further considers speed and angle of blow as the two main contributors to impact force. Kheisin, on the other hand, considers that impact force is a direct proportion of the mass of the vessel depending upon hull form and speed of ship.

Nogid states that ship's mass is important and shows that for a given ship's hull strength the speed of a vessel manoeuvring through small ice floes reduces to one quarter of this speed when manoeuvring in an infinite ice floe.

It is difficult to assess theoretically the effect of the ship's mass when striking an infinite or finite ice floe, but undoubtedly mass has some bearing. All authors seem to agree that impact force is directly proportional to speed.

The scantlings of the middle length should be based on the crushing strength of ice.

Hull form is of primary importance and highly concentrated loads seem unlikely, however, frame scantlings should be sufficient to withstand a fairly concentrated load placed at the most unfavourable point and shell plating should be designed on a uniformly distributed load equal to the crushing strength of the ice.

It can be seen that it is theoretically possible to evaluate formulae based on physical laws which take into account speed, mass, etc., but this approach does not appear very attractive and some statistical analysis of existing icebreakers seems unavoidable. It may be that such an analysis will produce certain similarities depending upon speed and mass which lend themselves to the adoption of simple formulae for determining principal hull scantlings, but this remains to be done.

Many of the Russian and Finnish icebreakers built in the past were to the classification of Lloyd's Register, but their class was withdrawn, at Owners' request, at a later date. Although, therefore, it is possible for Lloyd's Register to compare existing designs of icebreakers from the plans available, it has no extensive damage reports under known service conditions which can be studied and compared with the structures adopted, as is the case with ordinary merchant ships. On the other hand one can conclude that in the absence of evidence to the contrary, the structures have given satisfaction.

3. Structural Strength Criteria

(a) Shell Plating

Considerable theoretical work has been done on the theory of stresses in flat plates, generally of a highly mathematical content, the nature of which makes practical application difficult. Much controversy reigns over the methods and theories expounded and to date no single method has emerged which satisfies in a practical manner all the methods of olamping and loading.

It is known that a plate will withstand pressures considerably in excess of those required to just yield the plating, and as an example, if the bulkhead plating laid down in Lloyd's Register Rules for Steel Ships were based on elastic theory alone they would need to be considerably thicker than recommended.

It is convenient to consider a plate loaded in two distinct ways :-

(i) uniformly
(ii) concentrated

(i) Plating Uniformly Loaded

It is assumed that the backing structure is efficient in supporting the plating and all edges are clamped.

It is necessary first to define the criteria for permissible plate pressure and several investigators have laid down their own criteria ranging from initial yield with zero membrane stress to a maximum permanent set under pressure of 2t.

Reference to Timeshenko and Roarke assume elastic theory for plating with small deflections, and Hooke's Law holds for the material.

Simple beam theory does not apply in practice and it is generally recognised that yield stress should not be the criteria but rather permanent set. It is advisable to ignore membrane stress for thick plates. Before any simple formula can be propounded

it is advisable that such theory be combined with practical experience as there are too many variables and unknowns.

Reference is given at the end of this report to a paper by J.B. Caldwell, B.Eng., Ph.D., viz. "Notes on the Structural Design of Welded Ships" in which a series of curves is given showing the permissible lateral pressure in relation to breadth and thickness of plate, together with increase in pressure against yield strength of the material.

This curve, together with the various criterion for permissible pressure is reproduced in this report.

It can be seen that on a fixed spacing of stiffeners, the thickness increases as $\sqrt{\text{pressure}}$ and the permissible pressure increases as $\sqrt{\text{yield of material}}$. The curves range from initial yield with zero membrane stress to a maximum permanent set of 2t.

For a large icebreaker with a shell thickness of, say, 2" over a frame spacing of 16", 2t = 4" in 16" which is excessive.

Lloyd's Register's Rules for bulkhead plating are based on a plastic collapse with zero membrane stress in conjunction with a suitable factor of safety. (This is approximately curve DD_1 in Caldwell's paper).

In general it would appear reasonable to accept either curves DD_1 or DD_2 in Caldwell's paper. Curve DD_2 is for a permanent deflection of $0.2S\sqrt{(fy/E)}$ or S/150 for mild steel plates where S = stiffener spacing. This would be about 1/10th inch for the icebreaker plating.

This criteria could be taken as the normal working condition in ice and if desired, curve HH could be taken for the worst possible loaded condition.

It is worthwhile to consider the increase in permissible pressure by considering curves DD_2 and HH assuming S/t = 120.

From curve DD_2 pressure equals 10 lbs/in^2
From curve HH pressure equals 24 lbs/in^2
i.e. an increase of roughly 2½ times normal working pressure.

The curves assume an aspect ratio of 3.0 or more and exclude corrosion allowances.

The work done is generally confined to thin plating, but it is not thought that thick plating theory will be substantially different.

As stated above, the work has been confined to an aspect ratio of 3.0 or more. Timoshenko says that as the aspect ratio b/a increases the maximum deflection rapidly approaches that for a plate bent to a cylindrical surface obtained by making b/a equal to infinity. For b/a = 3.0 he states that the difference between the deflection of an infinite strip and the finite plate is about 6½%, from which it may be concluded that for comparison reasons when the b/a ratio is greater than 3.0 the calculations can be replaced by those for a strip without substantial error.

Lloyd's Register has made its own investigations into the effect of aspect ratio on plates subjected to a uniform pressure assuming plastic theory with zero membrane stress and assumes a maximum aspect ratio of 4.0.

The formula adopted by Lloyd's Register takes the following form :-

$$1.1 - \frac{s}{30S}$$

where s = stiffener spacing in inches
" S = length of stiffener in feet from support to support.

By inserting aspect ratios of 1.0, 2.0 and 3.0 into the formula, the percentage reduction in the basic plating formula reduces by 70%, 90% and 96.7% respectively.

Curves of aspect ratios from various sources are given in Fig. 1, together with the curve adopted by Lloyd's Register.

It can be seen that basically any formula adopted for plate thickness should be of the form

$$t = \text{Factor of Safety} \times \text{Stiffener Spacing} \times \sqrt{\text{pressure}} + \text{Corrosion Allowance.}$$

(ii) Plating under Concentrated Loading

For plating under the action of a concentrated load it is important first to decide what degree of concentration is to be legislated for.

It has been stated earlier in the report that a highly concentrated load is unlikely in ice, although it might be possible to have a "hard spot" in the ice of such a compressive strength that some concentration is possible.

Most of the theoretical and practical work done has been confined to ship's decks relating to the use of fork lift trucks.

Lloyd's Register has done some experimental work on the loads imposed by fork lift trucks and the effects on deck plating, and incorporated the results into its own Rules.

Other work has also been done by other authorities, but generally of a theoretical nature broadly based on Timoshenko.

The paper produced by Lloyd's Register is entitled "Investigations into the Use of Fork Lift Trucks on Board Ship" by W. Smith.

In the paper it is stated that overall stress will increase with stiffener spacing, but maximum stress (which is at the load) is not related to frame spacing. It further assumes a hypothetical yield stress of 25 tons/in^2 and compares the wheel loads with plating thickness and shows that within limits wheel area in contact has little or no bearing on plate stress.

It is important to realise that the paper works on loads and not pressures and that the criteria adopted is elastic with a hypothetical yield stress.

Although it is recognised that highly concentrated loads are unlikely in ice, the purpose of mentioning plating under concentrated loads in this report is to emphasise that the assumptions made for plating under concentrated loads are different to those made for plating under a uniformly distributed load. Under concentrated loads frame spacing has little or no

effect on stress and actual loads should be used, whereas for plating under a uniformly distributed load pressure should be used and frame spacing has a direct influence on stress. Furthermore, plastic theory with a specific load criteria of permanent set is assumed for plating under a uniformly distributed load and elastic theory in conjunction with a hypothetical yield stress has generally been assumed for plating under a concentrated load.

At the present state of our knowledge it is not considered advisable to try to rationalize the two approaches into one common basis.

(b) Framing Structures

The design of a structural framework may be considered from two criteria, namely

(i) plastic
(ii) elastic

(i) Plastic

With this approach the limit load is decided upon and a limit analysis made. With this assumption a more realistic limit stress can be obtained, and it also has the advantage that thermal stresses are eliminated in the plastic area, and full account is taken of the total energy under the stress/strain curve.

It is important to know what the limit load is, however, and set a factor of safety against this load. Plastic theory is often used in simple truss frameworks where the collapse load is more easily determinable and the interactions of connecting members more readily estimated.

The geometry and shape of each individual section has to be decided upon in addition to the normal elastic modulus. It is important that adequate lateral support be given to members to prevent twisting, for if the member twists it will probably fail at the yield point of the material.

Quite often model tests are carried out to determine the collapse load of a structure, and in

the more complicated structures full scale tests have been undertaken.

(ii) Elastic

This criterion is the more established one and limits the design to a predetermined stress, generally not exceeding the yield stress of the material. The stresses decided upon are generally the result of experience gained on similar structures which have given good service without yielding. As more knowledge is gained of a particular type of structure then are the design stresses increased accordingly.

Although it can be argued that the plastic approach is more logical and that once the collapse load is known, what loads go before it are irrelevant, it is important to know what the collapse load is. Certain ship structures lend themselves quite readily to plastic theory such as transverse watertight bulkheads where the hydrostatic loads are more easily decided. For an icebreaker, however, reference to the theoretical work given earlier in this report demonstrates that the forces acting during icebreaking are not known with any certainty and this, coupled with the quite complicated structure generally adopted in icebreakers would make it inadvisable to adopt plastic theory for determining principal framework structures.

Until further research is done on actual ships or full scale or model tests, it is suggested that elastic theory be adopted for frameworks with a design stress equal to the yield stress of the material.

4. Corrosion

The rate of corrosion for the hull of an icebreaker is far greater than for a normal cargo vessel. This is because of several reasons - the principal ones being as follows :-

(i) During the action of icebreaking large amounts of oxygen are released causing corrosion acceleration;

(ii) Salt concentrations will probably be greater particularly just under the ice at water level;

(iii) Stress concentrations accelerate corrosion locally particularly in way of the actual contact;

(iv) Abrasive forces are greater than for a normal cargo vessel.

According to an article written by Messrs. Rennie and Turnbull in the April 1964 edition of "Materials Protection", the Canadian icebreaking ferry "ABEGWEIT" showed a metal loss estimated to be about $\frac{1}{8}$" from 1948 to 1951 in the forward and after hull plates.

Early in 1953 another appraisal was made and this time pitting had occurred to an average depth of $\frac{1}{8}$" to 3/16", with a maximum depth of $\frac{1}{4}$". Weld metal had corroded to such an extent that the remaining weld material was as much as $\frac{1}{4}$" below the surface of the adjacent plates. This means that the corrosion rate for this vessel was between $\frac{1}{8}$" to $\frac{1}{4}$" over a seven-year period.

In conclusion it can be seen that corrosion is excessive on an icebreaker and a good margin has to be added to the bare designed thickness to counteract this.

5. Work in Progress

An elastic structural analysis is presently being studied for two large icebreakers, namely U.S.C.G. "GLACIER" and U.S.S.R. "MOSCOW", by use of computer.

The object of these analyses is to determine the structural effectiveness of two icebreakers designed from different structural philosophies, and to compare the advantages of one system with another.

The structural philosophy of "GLACIER" assumes a system of trussed frames of equal spacing and strength throughout the mid-length, supported by the longitudinal bulkheads and girders. The "MOSCOW" assumes a system of main frames, web frames and horizontal girders.

Because of the structural repetitiveness of "GLACIER" a two-dimensional analysis using IBM "Stress" program is being utilized. For "MOSCOW" it was considered that a two-dimensional analysis would not be a true representation of the structure under load and accordingly this is being analysed by the use of the three-dimensional IBM program "Fran".

It should be noted that for both these programs elastic theory is assumed. Bending moments, axial loadings and shear are produced by the programs and these are presently being analysed.

When calculating the inertias and moduli of the members an allowance was made for the attached shell, deck or bulkhead. The programs, however, assume an open grillage framework with all the load taken by the framework and none taken by the attached shell, acting as a support between bulkheads and decks.

In both cases a uniform pressure of 400 p.s.i. was assumed each with two conditions of loading 10 feet wide, the first condition being at or about the load waterline, the second condition being about 10 feet below the first.

A simple two-dimensional analysis has also been made of the Canadian icebreaker "LOUIS S. ST. LAURENT" which is based on the same structural philosophy adopted for "MOSCOW". The purpose of this analysis is to determine a simple basis for comparing the strength of other icebreakers of similar type but of different dimensions under fixed loads.

The results of the above analyses are the subject of a separate report and will be forwarded later.

6. Recommendations for Conclusion of Study

Depending upon the outcome of the structural analyses of "GLACIER" and "MOSCOW" a recommended structural philosophy will be given.

Comparisons of a general nature with some icebreakers built to Lloyd's Register Class will be made. These will be examined using relatively simple beam theory with certain assumptions made regarding fixity and will serve mainly to see if there is a general pattern in icebreaker structures.

An investigation will be carried out to determine the effect on structure with alteration in aspect ratio bearing in mind the practical limitations of construction. For this part of the study it would be desirable to have the spacings of decks and bulkheads of the U.S.C.G. proposal. From this study a structural effectiveness will be given with each change in aspect ratio based on stress levels, weight and cost of each modification. With regard to cost, this would be confined to the total weight of shell and structure within a finite area and the number of items and joints within this area and will, therefore, be only of a comparative nature.

The results of this investigation, coupled with the knowledge gained and information given throughout the study will form the basis of a recommended structural design for the U.S.C.G. polar icebreaker in association with the proposed general arrangement.

It is necessary for the U.S.C.G. to provide final ice load criteria.

L. J. Crighton
30/3/67
CMB.

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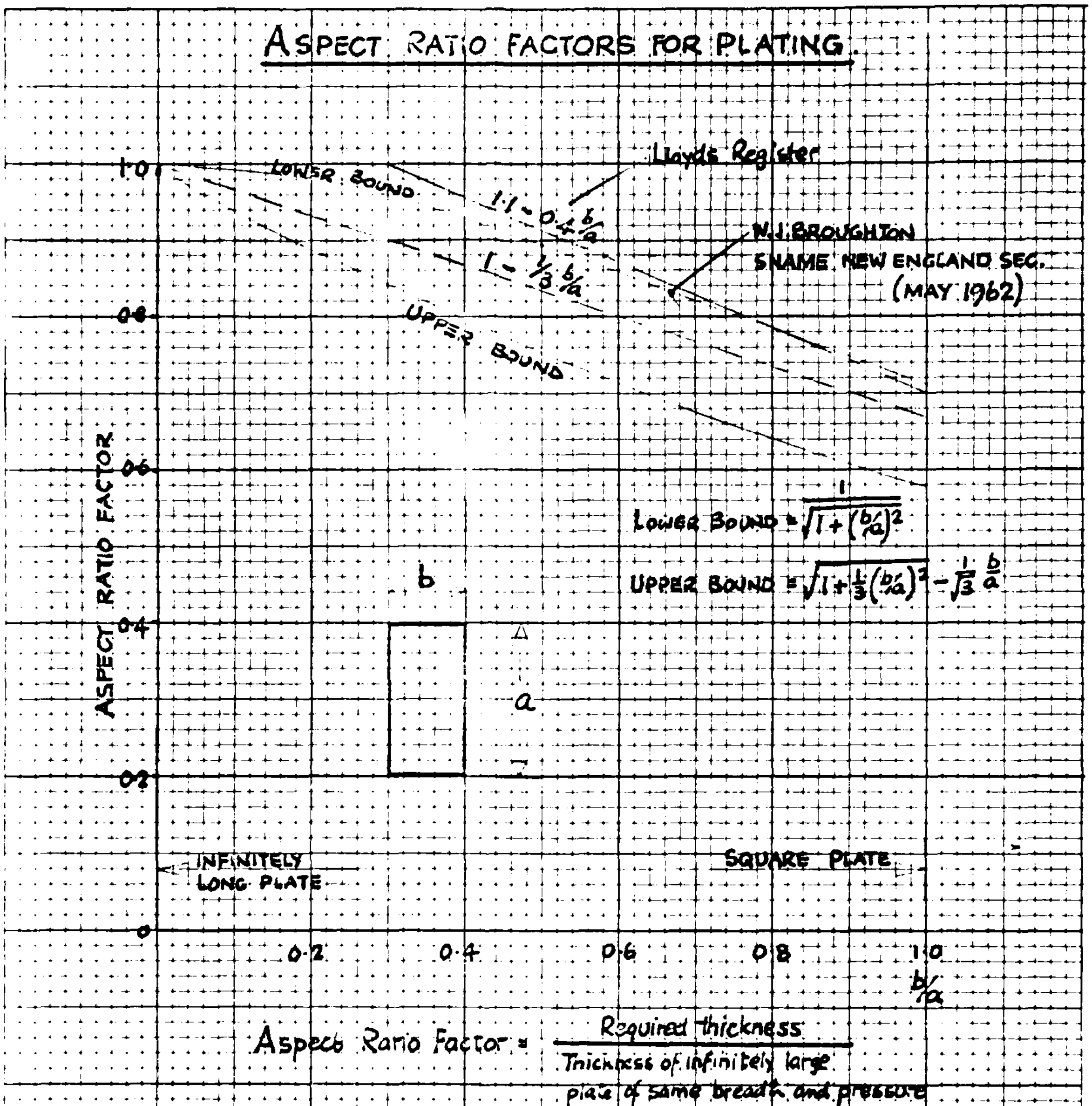
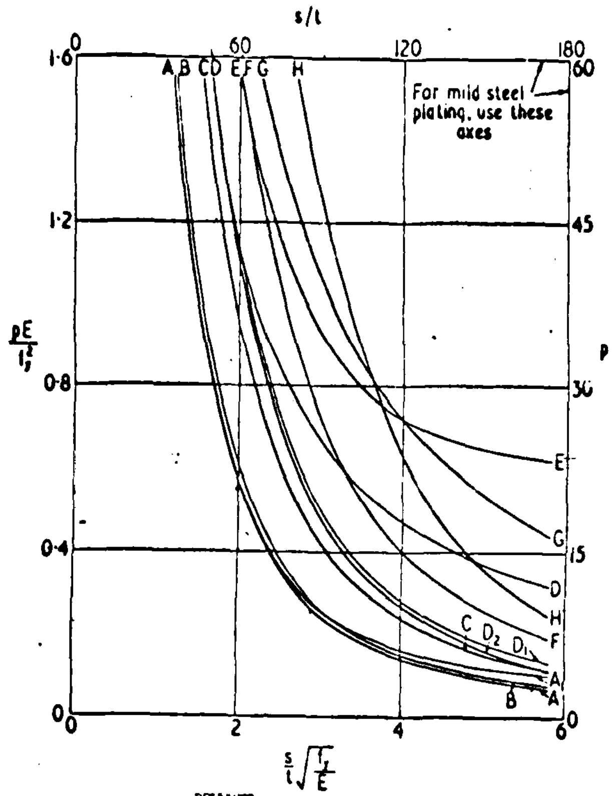


Fig. 1

Source - Lloyds Register of Shipping.



p = permissible ~~stress~~ pressure lb/sq. in.
E = Young's Modulus
f_y = Yield stress in tension, lb/sq. in
s = stiffener spacing, in.
t = plate thickness, in.
Corrosion allowance excluded
Plate length ≥ 3 × plate breadth

Fig. 5
Plating design curves

~~Table 1~~
Plating design methods

Curve (Fig. 5)	*Criterion for Permissible Pressure*	*Author*
AA	Initial yield—zero membrane stress.	
AA_1	Initial yield—full membrane stress.	
BB	Initial yield with fictitious yield stress. Empirical curve for tank bulkheads.	Vedeler[5]
CC	Initial yield with fictitious yield stress. Empirical curve for sub-division bulkheads.	Vedeler[5]
DD	Initial yield—full membrane stress. Assumed initial deflection $0{\cdot}2s\sqrt{f_y/E}$	Clarkson[6]
DD_1	Plastic collapse — zero membrane stress.	
DD_2	Max. permanent deflection $0{\cdot}2s\sqrt{f_y/E}$ Zero membrane stress.	Clarkson[7]
EE	Max. permanent deflection $0{\cdot}2s\sqrt{f_y/E}$ Full membrane stress.	Clarkson[6]
FF	Initial yield with fictitious yield stress. Empirical curve for sub-division bulkheads.	Admiralty
GG	Centre-line plastic hinge; or membrane stress $= \frac{2}{3}f_y$	Clarkson[6]
HH	Maximum deflection under pressure $= 2t$.	Young[8]

Reprinted for Dr. J.B. Caldwell's paper "Notes on Structural Design of Welded Ships".

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13. ABSTRACT

The four reports, Nos. 5047,5048,5049 and 5051 undertake a comparison of the structure of the U. S. Coast Guard icebreaker GLACIER and the Russian icebreaker MOSCOW. The GLACIER was analyzed by the STRESS Computer Code, while the MOSCOW analysis was performed using the FRAN Code. The results of the study showed the MOSCOW grillage structure to be superior to the GLACIER truss structure both in regard to elastic strength and plastic collapse under collision penetration.

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